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Experimental Analysis of the V-Charge Variable Drive Supercharger System on a 1.0 L GTDI Engine

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Abstract:

A compound charging system which pairs a turbocharger with a supercharger seems to be a potential trend for future passenger car gasoline engines as the strength of both could be enhanced and the deficiencies of each could be offset. The use of a fixed-ratio positive displacement supercharger system on a downsized turbocharged gasoline engine has already appeared in the market. Although such systems can achieve enhanced low-end torque and improved transient response, several challenges still exist.

An alternative solution to the fixed-ratio positive displacement supercharger is the V-Charge variable ratio centrifugal supercharger. This technology utilizes a Torotrak continuously variable transmission (CVT) coupled to a centrifugal compressor for near silent boosting. With a wide ratio spread of 10:1 and rapid rate of ratio change the compressor speed can be set independently of the engine speed to provide an exact boost pressure for the required operating points, without the need to recirculate the air through a bypass valve. A clutch and an active bypass valve can also be eliminated, due to the CVT capability to down-speed, thus improving the NVH performance. This paper will, for the first time, present and discuss the V-Charge technology optimization and experimental validation on a 1.0L GTDI engine to achieve a better BSFC and transient response over the turbo only and the fixed-ratio positive displacement supercharger solution. The potential for the V-Charge system to increase the low-end torque and enable a down-speeding strategy is also discussed.

Keywords: V-Charge, Variable Drive, Supercharger, Downsized, Gasoline engine

1 Introduction

Engine downsizing, which is the use of a smaller-swept-volume engine to provide the power of a larger one, is widely accepted as one of the most viable solutions to address the fuel economy and environment issues facing passenger car gasoline engines [1-3]. Reduced pumping losses, improved gas heat transfer and better friction condition thus shifting the engine operating points into a more efficient area, are the major reasons for improved fuel efficiency in the frequently-used areas of low-load engine operation. The rated power and torque are conventionally achieved by the adoption of turbocharging [4-5]. However, turbocharged engines characteristic insufficient low-end torque and poor transient response (so-called 'turbo-lag'). This might lead to a lower transmission gear selection to maintain a target vehicle acceleration requirement, which would sacrifice some fuel economy that is initially achieved from downsizing. In addition, the driveability of a turbocharged passenger car is usually degraded compared to a naturally aspirated counterpart, which in a sense trades some 'driving fun' for the fuel efficiency. Supercharged engines, especially in a compound charging configuration, might address the aforementioned issues of turbocharged engines due to the introduction of a forced induction device which, unlike turbocharging, is directly connected to the engine crankshaft. Although supercharged engines are not as fuel-efficient as its turbocharged counterpart at the same engine operating point, in a real world, supercharged vehicles due to their capability to further downsizing without largely affecting the driveability and to enable a down-speeding strategy from an enhanced low-end torque response may achieve better fuel efficiency and improved driveability simultaneously [6].

Currently most of the supercharged passenger car gasoline engines are fitted with a fixed-ratio positive displacement device, among which Volkswagen 1.4 TSI [7], Volvo T6 [8-9], Ultraboot [1, 10-13] are three examples that are using both a turbocharger and a supercharger (all are Eaton type) to provide all the necessary boost. They have all demonstrated better performance compared to a similar-size turbocharged rival. However, this conventional supercharger system

may not be fuel-efficient due to the need to recirculate air through a bypass valve and to run the compressor at excessive speeds for much of its operation, causing inevitable mechanical and flow losses. To be more specific, the use of a fixed-ratio supercharger in a 2-stage system usually requires a high drive belt ratio to achieve the desired boost pressure at low engine speeds. This approach will generally necessitate the use of a dis-connectable clutch and an active bypass valve to avoid over-speeding the supercharger at higher engine speeds and in order to reduce the parasitic losses to an acceptable level at part loads, which will inevitably increase cost, complexity and package significantly. Such a supercharger system is also well known to cause increased NVH issues [14].

An alternative to the fixed-ratio positive displacement supercharger is the V-Charge variable ratio centrifugal supercharger. This technology utilizes a Torotrak continuously variable transmission (CVT) coupled to a more-efficient centrifugal compressor for near silent boosting. With a wide ratio spread of 10:1 and rapid rate of ratio change (within 360 ms) the compressor speed can be set independently of engine speed to provide the exact amount of boost pressure for different engine operating points, without the need for wasteful bypassing. A clutch and an active bypass valve can also be eliminated, due to the CVT capability to down-speed which also mitigate the issues of cost, complexity and package and at the same time greatly improve the NVH situations [15].

2 Brief Review of Relevant Research

In general, there are two main types of superchargers defined according to the method of gas transfer: Positive displacement and aerodynamic compressors [16].

For automotive applications, positive displacement units are generally referring to the Roots and twin-screw types (the latter incorporating internal compression and therefore correctly termed a compressor, the former achieving all compression externally and therefore being properly a 'blower') [17-18]. These devices offer a relatively constant boost characteristic since they pump air at a fixed rate relative

to the engine speed and supercharger size (thus lower speed machine) [19] and they may be a more easily integrated device as only a simple drive system is needed. **Figure 1** shows an Eaton TVS R-Series supercharger view to indicate the integrity of this type of positive displacement unit.

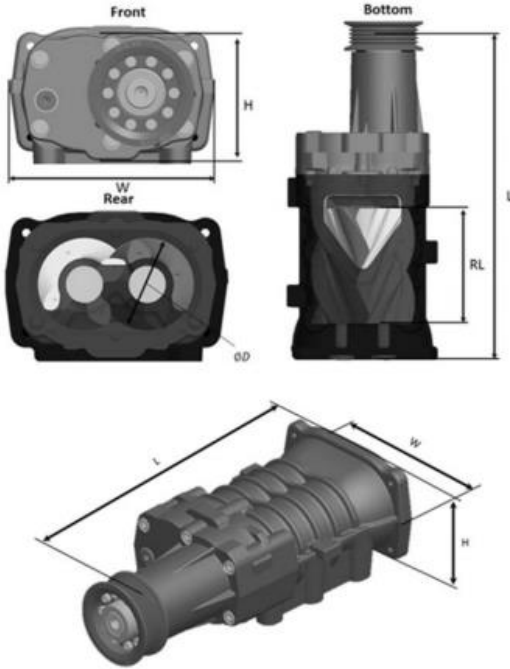


Figure 1. Eaton TVS R-Series supercharger view. [20]

The type of aerodynamic compressor for automotive applications is usually a centrifugal device as it is generally more efficient, smaller and lighter than their positive displacement counterparts. Similar with a conventional turbocharger, only a low boost is supplied at low rotational speed which is not ideal for an automotive engine which uses far more of its operation. In addition, the pressurization requires high tip speeds and therefore in order to provide sufficient boost, centrifugal superchargers usually need to be driven by some form of step-up gears, which inevitably incurs some additional mechanical losses [14].

Low-end torque enhancement, transient driveability improvement and low load parasitic loss reduction are deemed to be the three potential development directions for a mechanical supercharging system [6]. Since the adoption of a CVT to decouple the supercharger speed and the engine speed is able to augment the low-end torque [21], improve the transient behavior [14, 22-23], and potentially eliminate the low load parasitic losses [24-25], a simulation and

experimental study of a CVT driven supercharger device is in line with the potential future development. In addition, for a purely mechanical supercharging system transiently, a large amount of inertial torque will incur which might either degrade the driveability or affect the package or the durability of the CVT adopted. In this context, low inertial centrifugal compressor may be preferred. The Torotrak V-Charge system is capable of altering CVT ratio rapidly and it can be fitted with a centrifugal compressor, thus is now considered to be a potential solution to address the fuel efficiency and driveability issues [14].

E-boosting, which is realized by electrically driving a compressor (usually centrifugal), is also able to decouple the boost process from the engine operating point. It is usually coupled with a conventional turbocharger, and the air mass flow and pressure ratio provided by the electric supercharger is essentially free if provided from the stored recovered energy [26]. This either results in an engine system with more produced brake torque or translates to better fuel economy when a fixed engine torque is required [27]. Compared to the V-Charge solution, however, the standard 12V lead-acid battery system might not be able to provide the same level of transient performance [15] and can only be running continuously at high boost pressures and high mass flow rates for a limited period of time, due to temperature limits and battery capacity [28].

3 Scope and Objectives

This paper will detail an investigation to look at alternative boosting systems applied to a Ford 1.0 L EcoBoost engine [29], in order to achieve an enhanced target torque and power curve (see **Figure 2**). A larger turbocharger is fitted to match the power at the high end and the supercharger system is utilized to not only improve the transient performance but also significantly augment low-end torque due to the fact that a large torque difference between high and low engine speed can also cause a perceived turbocharger lag feel during vehicle launch even if the boost system response is more than adequate [27]. The objective is to assess the performance of a centrifugal-type supercharger system driven via a Torotrak continuously variable transmission against the turbo only and the

fixed-ratio positive displacement supercharger solution.
Figure 3 shows V-Charge system design and testing

workflow that will guide the simulation and testing procedure later.

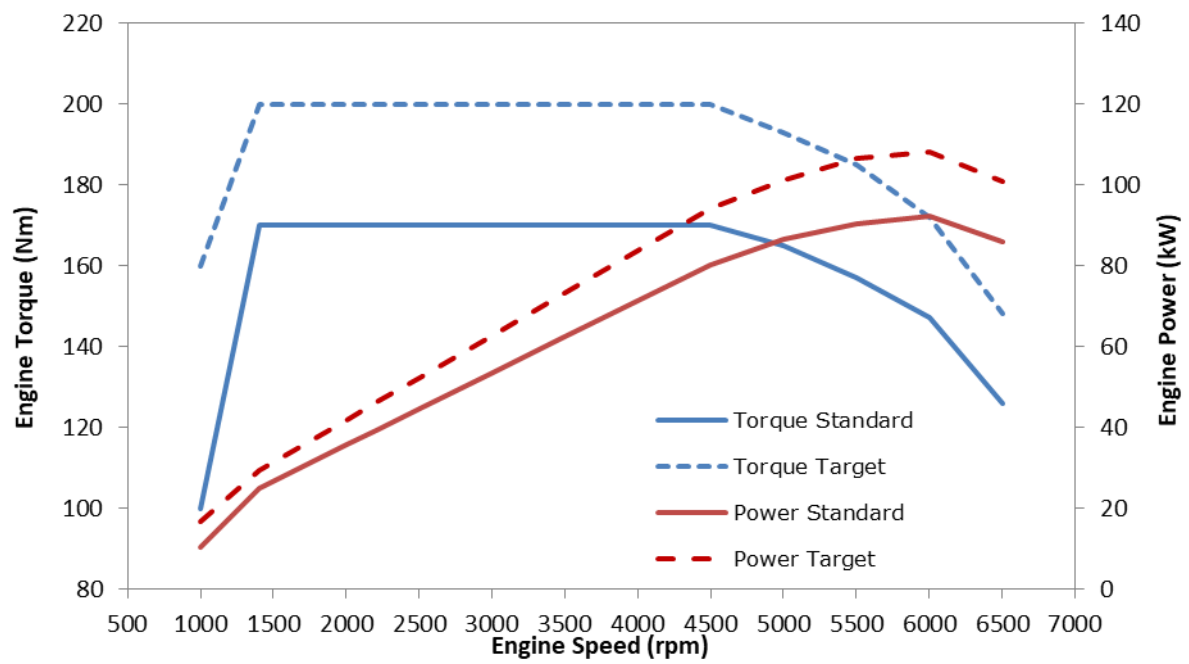


Figure 2: Original standard and targeted engine performance

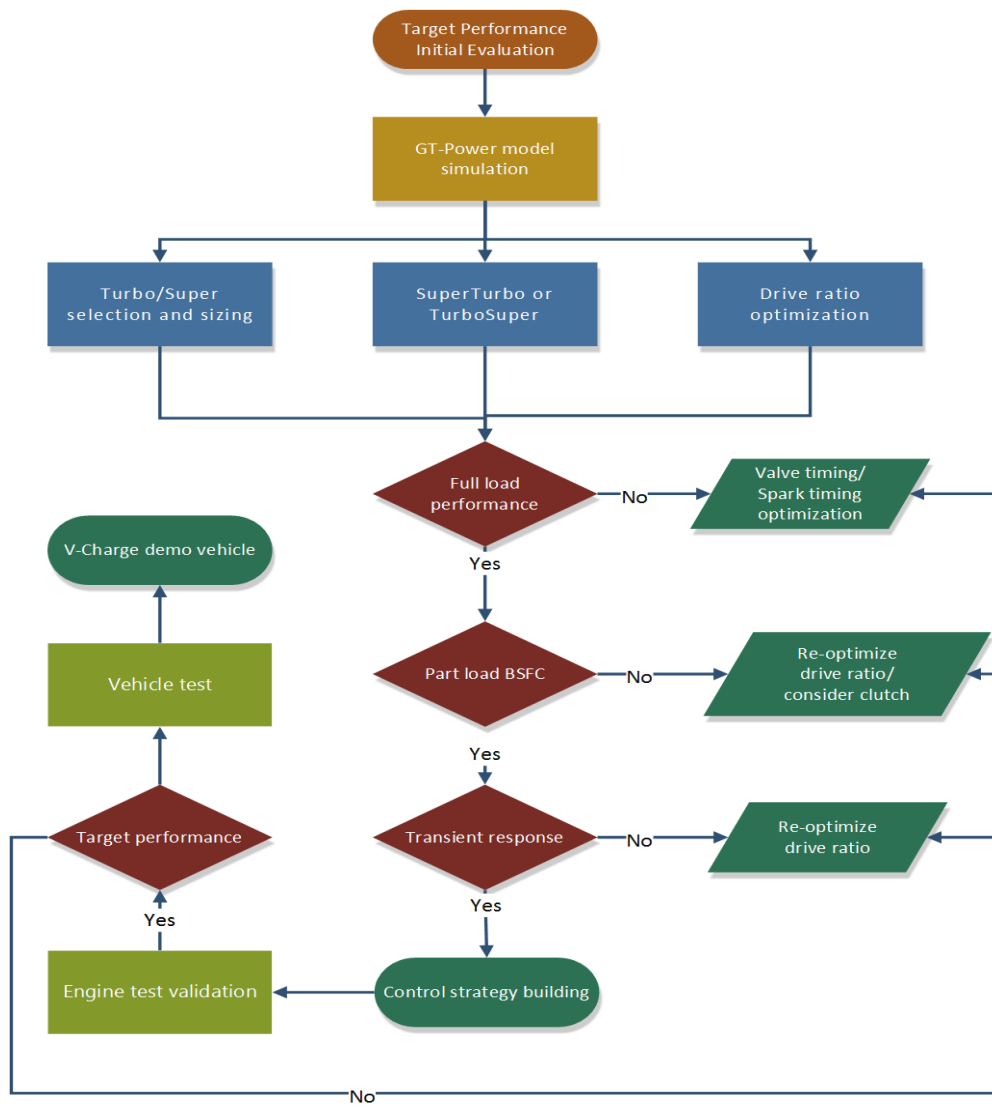


Figure 3. V-Charge system design and testing workflow

4 Preliminary Simulation Results

Figure 4 shows the integration of the supercharger options within the engine boosting system, as a supercharger followed by turbocharger arrangement – SuperTurbo. Both SuperTurbo and TurboSuper configurations were considered in simulation, with very similar overall results achieved. But it might be noted that there is no intercooler planned to install between the two boosting devices and different charging arrangement will result in the operating points of both compressors ending in different regions of the corrected compressor map. This might either affect compressor surge margin performance or influence the supercharger power request. In general, from the perspective of fuel efficiency, TurboSuper will give a better result than SuperTurbo as the turbocharger compressor at the low pressure stage will generally be

more efficient and will be able to produce more PR than when the turbocharger compressor at the high stage, if assuming the drive transmission efficiency and turbocharger power is fixed. However, if the supercharger is put at the high stage, a better transient response is anticipated as it is closer to the engine cylinder. For ease of packaging and boost pipework routing on the engine hardware, a SuperTurbo configuration has been chosen and pursued in the following test phase.

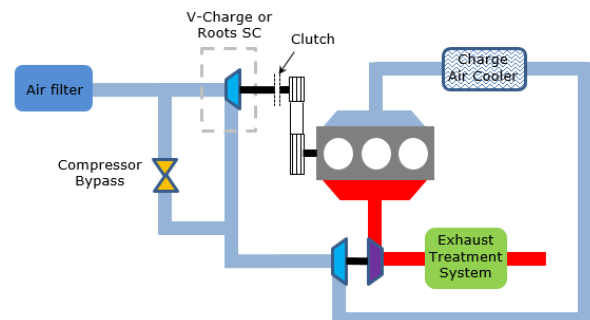


Figure 4. Schematic of the proposed engine equipped with the V-Charge system

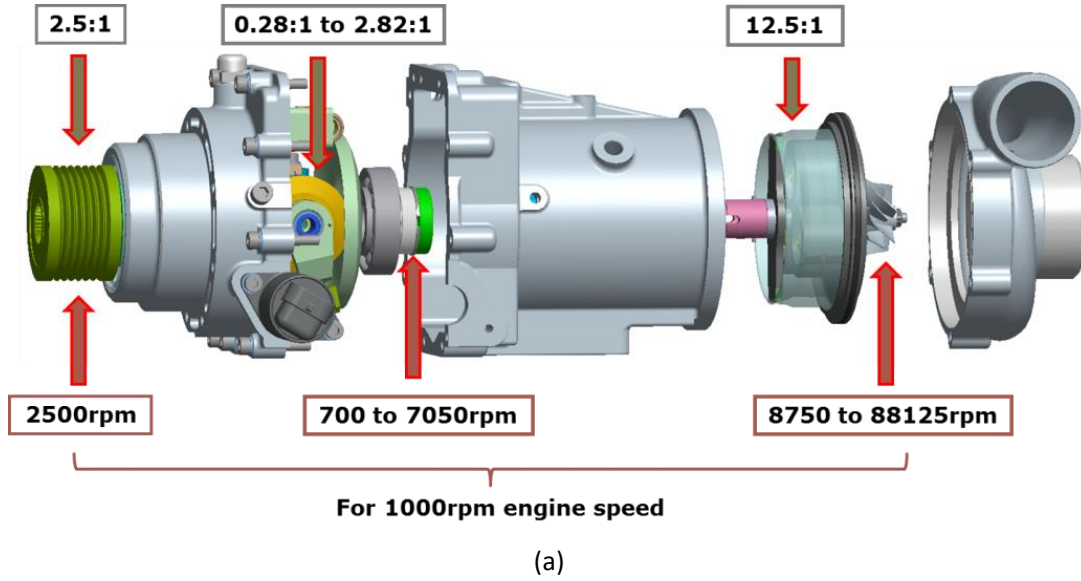


Figure 5. V-Charge variable ratio supercharger – ratios and speeds

There are two types of compressor that were considered in the simulation phase. A conventional centrifugal compressor and a novel compressor that is supplied by Honeywell. The latter one features the same pressure ratio (PR) at lower rotational speed but characteristics lower maximum PR (maximum rotational speed is approximately halved), compared to the conventional compressor wheel. It gives a bearing loss improvement due to reduced speed, but a large BMEP enhancement would lead to a high PR request that might not be achievable for the novel compressor.

The compressor size was optimized for low flow rates and in order to enable a seamless handover between

the two boosting devices in operation (to avoid torque interruption which would give an undesirable drive feel in the vehicle), a sufficient map overlap between the V-Charge compressor map and the main turbocharger map was determined.

The drive ratio which includes both the step-up and the epicyclic ratio were optimized considering the trade-off between the transient performance and the part-load BSFC. As there is no clutch fitted with the V-Charge system, a larger drive ratio will result in deteriorated fuel consumption but enable a faster tip-in response, vice versa.

After optimizing the compressor size and the drive ratio (the ratios and the exemplary speeds seen in **Figure 5**), it was shown in the simulation that the BSFC at full load was improved by approximately 1% due to its capability to eliminate the recirculated mass flow around the supercharger resulting in reduced mechanical and flow losses. The BSFC performance could be further improved by another 1% if a Honeywell novel compressor was used which is mainly attributed to the reduced transmission losses and increased compressor isentropic efficiency.

Under part load, it was demonstrated that a clutch might not be necessary for the Torotrak V-Charge system and only a minor decline of the part load BSFC performance was observed if a clutch was not used (the V-Charge system is constantly connected). For the novel compressor, the part load BSFC will have slightly degraded BSFC performance compared to the conventional compressor due to the higher PR at fixed compressor rotational speed. However, as the transmission ratio could be reduced for the novel compressor configuration without largely affecting the engine's transient performance, the part load BSFC could be reduced below the conventional counterparts.

In transients, the simulation results suggested that the Torotrak V-Charge system with a conventional compressor could provide better transient performance compared with the fixed-ratio positive displacement supercharger rivals. The novel compressor configuration can provide even better transient performance at low engine speed while maintaining a similar performance response at high engine speed.

5 Control Strategy Implementation

The control strategy was also initially constructed in the simulation phase. In order to easily access the engine sensor measurements for use in the two-stage boosting control strategy, an aftermarket OpenECU was used in combination with the OEM development engine ECU. However, in a production solution the V-charge and the wastegate control would be achieved on the OEM ECU, via PWM signal to the actuator that controls V-Charge speed and wastegate position.

The proposed control sequence is basically composed

of a driver input which is the pedal position and how the input affects the actuator actions in the engine system. For details, see **Figure 6**. The ECU will first receive the pedal position and engine speed signals from the driver, and will then calculate the demand total boost pressure, followed by the determination of the boost split (feed-forward loop). Finally, a feedback control loop is required to correct the demand boost pressure to the target, under different boundary conditions (or, in other words, a different altitude) or component ageing.

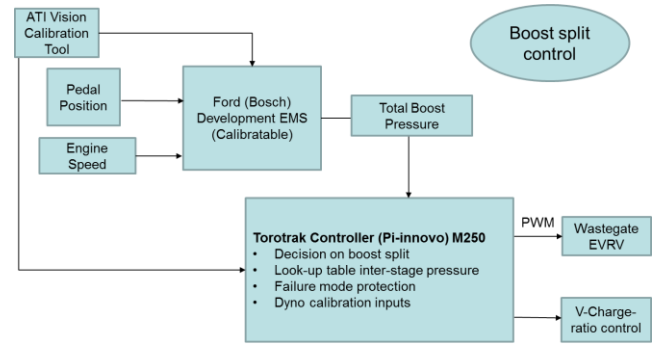


Figure 6. Boost split control module

In order to build a robust control strategy and gain some knowledge of tuning the behaviour of this two-stage boosting system, a GT-Power model was initially used to simulate the gas dynamic response, while the control strategy was generated from Simulink. This approach was also beneficial later, when a near-to-complete control strategy was transferred from Simulink to an open ECU developer platform in the experimental phase, due to the fact that the open ECU that is adopted in this project can be compiled in a Simulink environment.

The proposed V-Charge speed control module (see in **Figure 7**) is made of several sub-systems, including a feedforward steady-state look-up table and a feedback loop on intake manifold pressure.

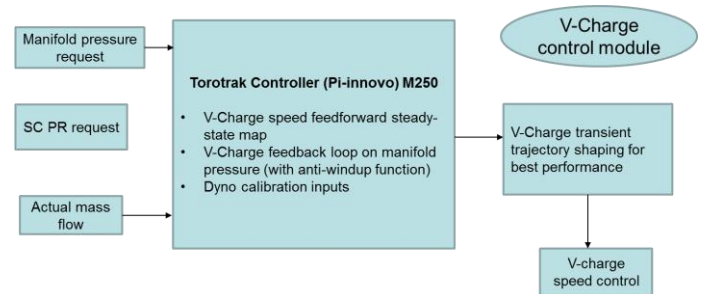


Figure 7. V-Charge speed control module

The feed-forward loop, which is basically a look-up table, is designed to help the controller to achieve the target quickly and robustly. This map is characterised by the strategy of boost split. It might be noted that the control strategy currently adopted in this work is one of minimizing mechanical boosting in the interests of efficiency. The aim is to achieve the highest possible utilization of the turbocharger, and the supercharger only assists when the turbocharger is unable to provide adequate pressure ratio at low engine speeds and during transient events.

The feedback control includes an anti-windup function and a gain scheduling strategy. The feedback controller here, essentially, has two targets here: adjusting the V-Charge speed so that the total boost pressure can be achieved if the feed-forward loop is not working precisely (due to component ageing or different boundary condition); and detecting a transient signal in order for the supercharger to 'pre-boost' the engine system.

It might be worth noting that two look-up maps (i.e. upper and lower V-Charge speed rate limiter) were used to optimize the transient trajectory shaping for the best V-Charge performance, which was from the perspective of physical response behaviour, and also the driveability consideration. At the time of writing, it is found in simulation and test that the largest achievable CVT change rate might not be appropriate to implement during a transient event, due to the 'dip' phenomena, as there may be too much acceleration torque generated (see **Experimental Results**). In addition, whether the torque dip (if there is any) could be 'felt' by the driver is another question, which also needs to take into consideration the damping effect of the whole powertrain system (and is thus beyond of the scope of this research).

In order to accurately control the boost split in the two-stage system, the standard closed-loop pneumatic mechanism was replaced by a 'smart' pneumatic wastegate mechanism with electronic control. **Figure 8** shows the wastegate control module that was proposed

for the V-Charge project. It is, like the V-Charge speed control discussed above, was comprised of two main sub-sections of controlling: feed-forward and feedback loop. There is also an anti-windup loop in the wastegate control module. Note that the PI controller input for the wastegate control is the turbocharger pressure ratio difference between the target and the actual, and that for the CVT ratio control is the total boost pressure difference between the target and the actual. This is scheduled in order for the two interdependent PI controllers not to fight each other.

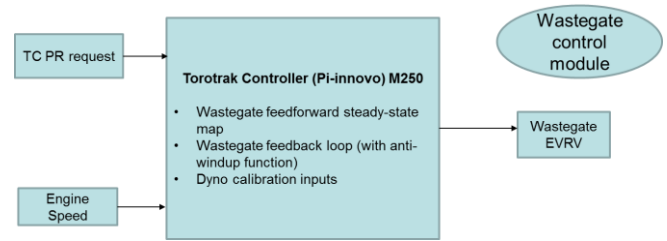


Figure 8. Wastegate control module

6 Experimental Setup

The experiment validation was carried out in an engine test facility at University of Bath (see **Figure 9**). A comprehensive instrumentation and measurement approach ensures accurate verification of system performance. The system set up and measurement details are shown in **Figure 10**. In the facility, the ECU calibration software ATI Vision is communicated with the host system CP Cadet via an ASAP3 link. ATI Vision is also communicated with the ECU via a CAN Calibration Protocol (CCP) and selected channels can be transferred to CP Cadet for the easiness of record and monitoring.

The Torotrak V-Charge system, which is comprised of a pulley step-up gear, a CVT mechanism and an epicyclic gear as shown in **Figure 5**, was directly connected to the engine crankshaft via a conventional micro-V belt, and in this proof of concept installation is achieved with a separate additional pulley. This is mounted alongside the standard fit FEAD pulley as shown in **Figure 11**. For a production integration the supercharger could be driven by an upgraded FEAD belt, reducing the losses associated with an additional belt driven system.

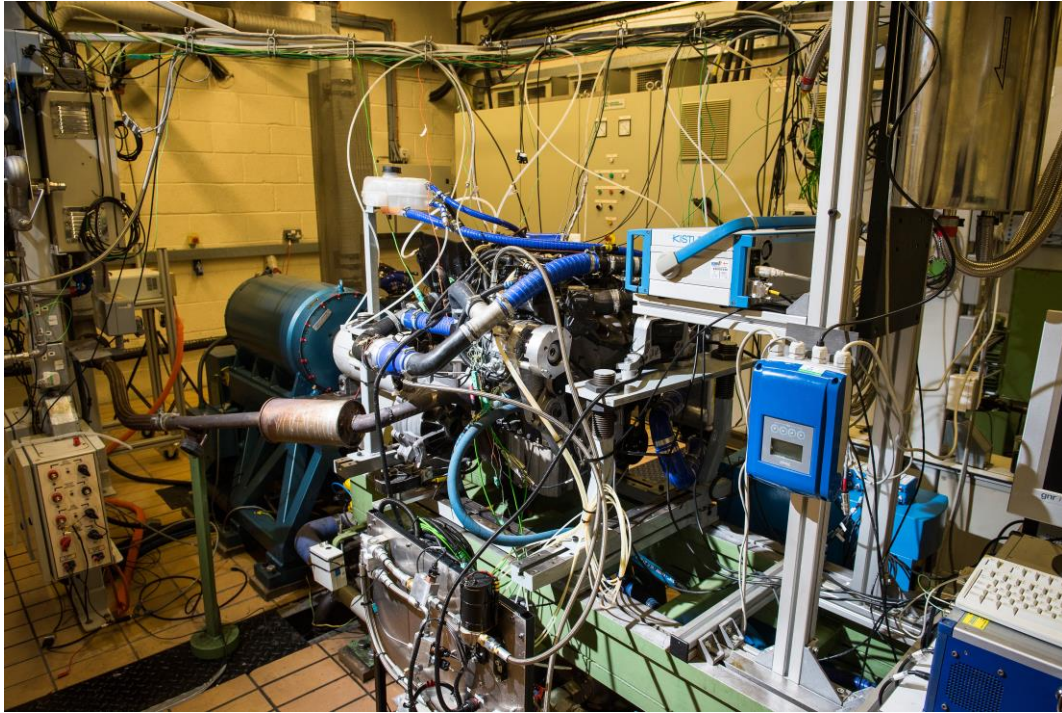


Figure 9. Test cell at University of Bath

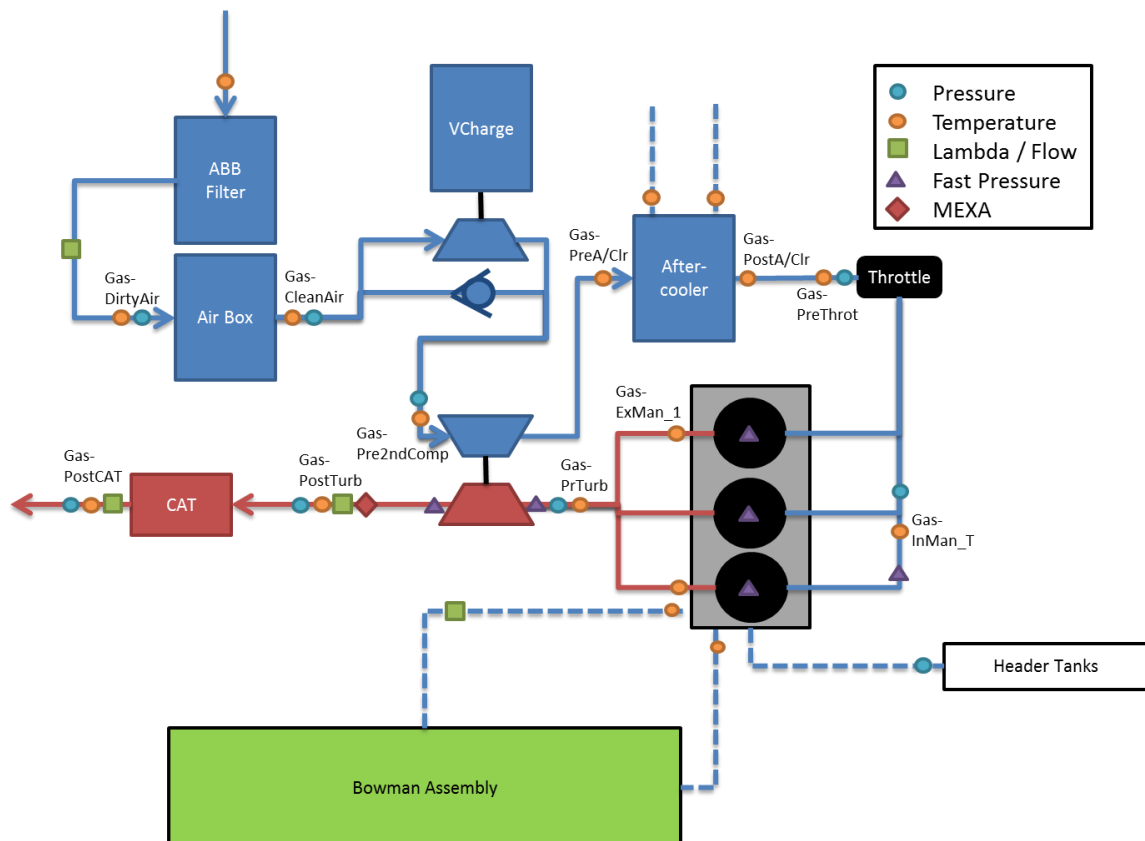


Figure 10. System schematic and measurement locations

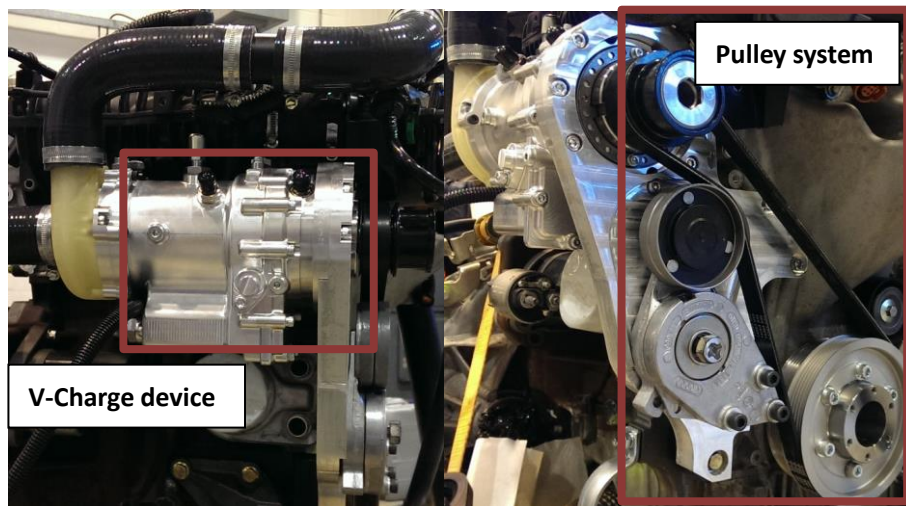


Figure 11. V-Charge system installation

A check valve seen in **Figure 12** was mounted around the V-Charge compressor to act as a passive bypass valve in order to bypass the compressor when the V-Charge compressor is not able to provide the required mass flow rate.

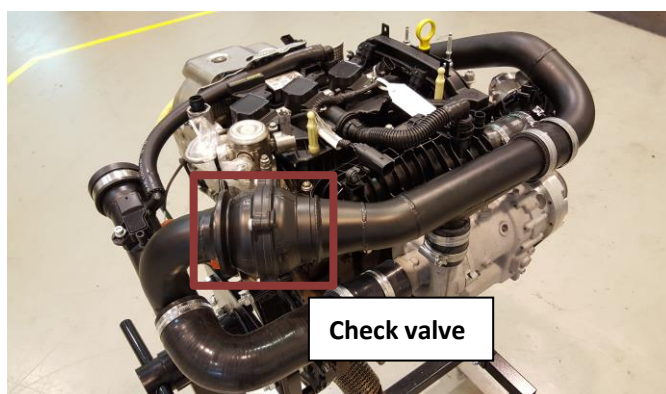


Figure 12. Check valve configuration

In order to have wide operating range and precise control of the engine boundary conditions, the cooling circuit for the engine coolant and oil were replaced with an external water-to-coolant heat exchanger. In addition, an aftercooler was replaced by a water-to-air heat exchanger. Therefore the temperatures of the engine coolant, oil and engine intake air could be controlled by varying the water flow rate in each heat exchanger. In this test, before logging the test data, the engine coolant, oil and aftercooler temperature was kept within $\pm 1^\circ\text{C}$ of the following set-points, 90°C , 100°C and 45°C .

7 Experimental Results

Full Load performance:

The simulation and verification test activities have initially focussed on a modest increase of maximum steady state output torque, from circa 170Nm to 200Nm (see **Figure 2**).

The simulation phase suggests that despite the V-Charge system having higher mechanical losses with its integrated CVT, it can eliminate the recirculated mass flow around the supercharger resulting in lower power consumption compared to the positive-displacement supercharger system. However, the fuel consumption also depends on the matching of the engine and the boosting systems. For example, if the drive ratio of a positive displacement solution is only determined by the full load target, it may have better fuel efficiency compared to the V-Charge system as there is no recirculated mass flow generated. In addition, at just boosted condition, the V-Charge will have the largest BSFC advantage over the fixed-ratio positive displacement counterpart as the V-Charge can give the exact boost requirement while the target boost pressure of the fixed-ratio positive displacement solutions has to be controlled by a bypass valve, thus generating more mechanical and flow losses.

In light of the pursuit of higher BMEP's to facilitate more aggressive downsizing and fuel economy benefits, an additional stretch torque target of 240Nm was

specified, that reflects the output of the current Ford 1.5L 4 cylinder Ecoboost engine. This steady state target for the 1.0L Ecoboost engine equates to more than 30bar BMEP.

As can be seen in **Figure 13** and **Figure 14**, the test results for the V-Charge system demonstrates that the stretch torque target is achieved across much of the engine operating range. This is realised within safe combustion limits with headroom to potentially increase output further still.

The original target output at 1000rpm is not quite reached for steady state operation, with 145Nm achieved vs a 160Nm target. Transient torque capability at this speed is ~200Nm before falling back to 145Nm after a few seconds. This appears to be caused by the onset of turbocharger surge (indicated by the turbocharger compressor operating points shown in **Figure 15 (a)**), and may be remedied by adopting a TurboSuper arrangement rather than the SuperTurbo

configuration pursued in practical testing. Optimising turbocharger trim may also help with this limitation.

It should be noted that the superior transient response of the V-Charge equipped engine particularly at lower engine speeds will result in significantly improved driveability over the standard 1.5L turbocharged unit, something that the steady state torque graph cannot reflect.

The level of V-Charge engine full load steady-state performance (together with improved transient behaviour which will be discussed later) also alleviates the requirement of scavenging: a common approach to improve low-end torque of a direct inject turbocharged gasoline engine [30]. This will either aid in minimising the engine-out emissions [28] or reduce catalyst exotherm when operating with stoichiometric exhaust [31].

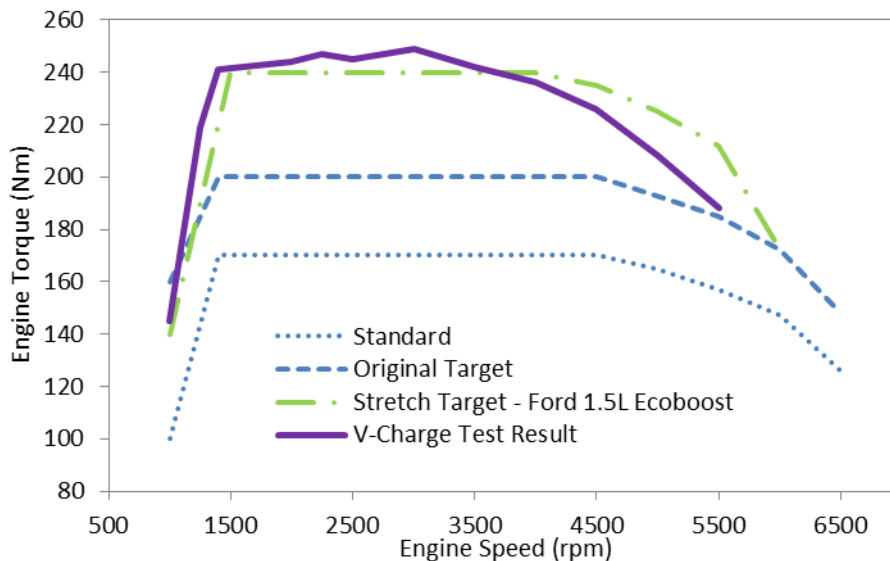


Figure 13. Engine torque output comparison

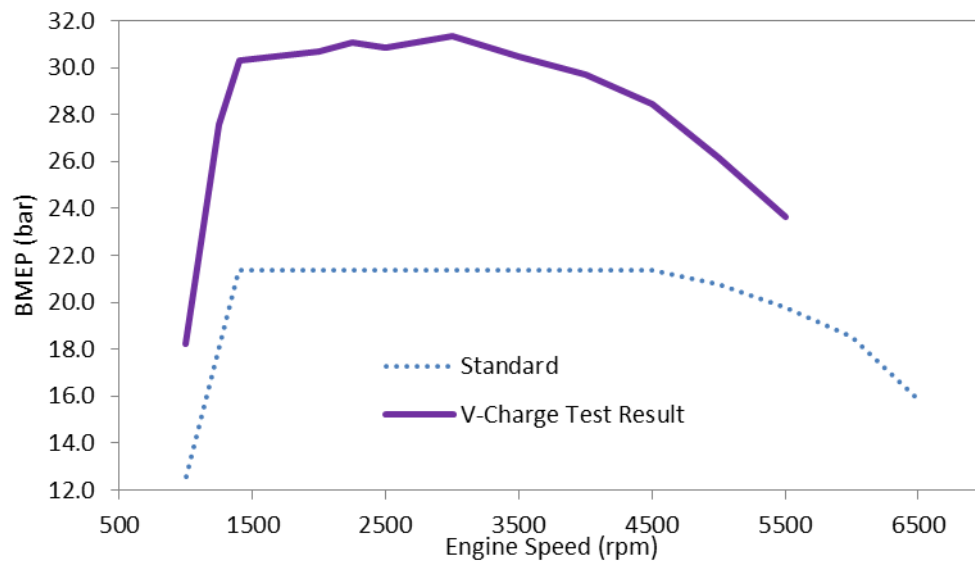
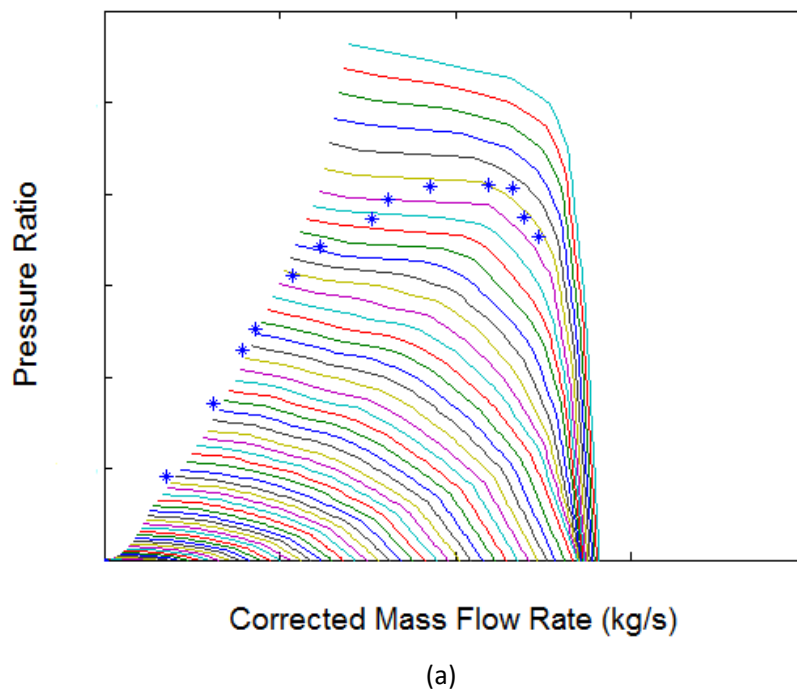
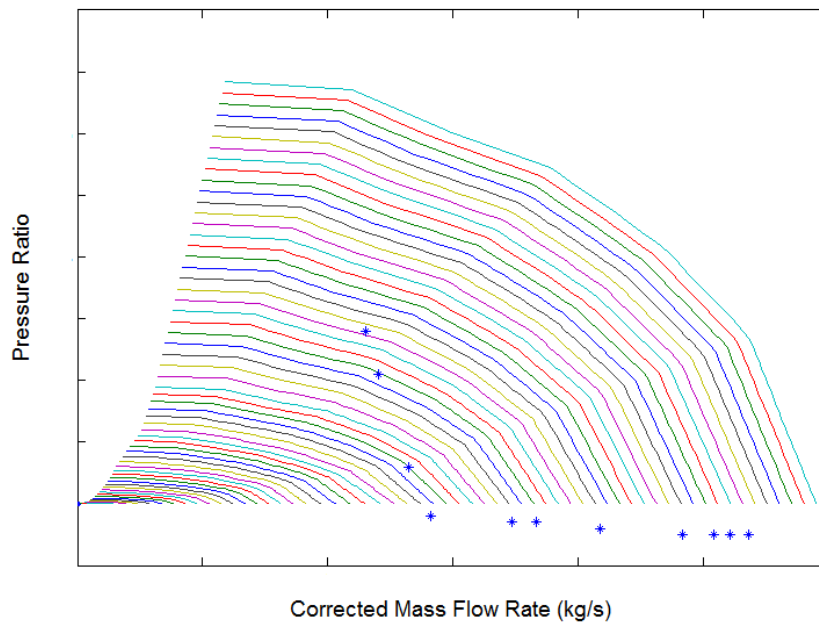


Figure 14. Engine BMEP comparison





(b)

Figure 15. Steady-state turbocharger compressor (a) and V-Charge compressor (b) operating points from 1100RPM to 5500RPM

Low load performance:

As mentioned in the work [14], the compressor of the Eaton configuration usually needs to be disengaged at low load, especially when a large drive ratio is utilised. However, the V-Charge system could trade some fuel efficiency at low load for better transient behaviour by constantly connecting the compressor to the engine crankshaft with its minimum ratio.

As there was a passive bypass valve installed, only one-way flow is allowed. If the CVT ratio is sufficiently low, there would be some pressure drop across it that would force the bypass path to open, which will then result in the supercharger PR to be around 1.

According to the test data that sweeps the engine torque from very low to the largest within the NA line using the minimum CVT ratio, it can be seen that the supercharger PRs were all around 1 that indicates that at the minimum CVT ratio, the supercharger could not generate sufficient pressure ratio without the assistance of the bypass valve, consuming more power.

It is known that if the PR and the speed of a compressor are about the same, the power consumed to drive the compressor should also be similar. Thus, it is safe to say

that for different engine operating points at the same engine speed, the parasitic losses are around the same. As there was no mass air flow sensor mounted in the supercharger path, the power consumed will be calculated in the validated simulation model in this work.

The simulation showed that at 1000RPM, only approximate 60W was wasted to constantly connect the supercharger compressor to the crankshaft, the ratio of the consumed power to the engine power being around 2% at low engine load (below 30N*m). At higher engine speed, a similar situation was observed. **Figure 16** shows the engine operating points for a C-segment vehicle on a WLTP driving cycle with the calculated weighted Minimap points. The test results with these points can be seen in **Figure 17**, with the maximum BSFC deficit around 5.5% and descending with higher engine torque. It should be noted that the BSFC deficit was calculated by the configuration of the V-Charge with its minimum CVT ratio compared with the counterpart with the V-Charge pulley off. The discrepancy of the simulation and test might be the underestimation of the pulley parasitic loss.

In order to reduce these parasitic losses, a novel

approach was proposed that just takes the bypass valve out of the system. This so called ‘wind milling’ effect will force the PR of the supercharger to be below 1 at low loads, resulting in the energy flow to be reversed, in order to offset some of the parasitic transmission losses. However, if the V-Charge control calibration is considered, the use of a passive bypass valve will be beneficial. This is mainly due to the fact that, at low load within the throttled region, the configuration with a bypass can constantly keep the CVT ratio at its minimum, while for the system without a bypass valve, the CVT ratio might need to be tuned to generate required vacuum (the parasitic losses will also be increased along with the durability of the CVT) [32].

Also it should be noted that compared to the standard engine configuration with a smaller turbocharger, the V-Charge system, featuring a larger one, is potentially able to reduce its part load BSFC by approximately 2% due to its reduced backpressure [27]. Besides, due to the improved transient behavior, the V-Charge engine also

facilitate a better pumping work due to the fact that the throttle can be set more open for the V-Charge engine at part load whilst for a turbocharged counterpart in order to enhance the transient performance the throttle are usually closed more than it needs to be to pre-spin up the turbocharger.

Compared to the fixed-ratio positive displacement counterparts which often have to fit an active bypass (and a clutch) to reduce the parasitic losses at part load, the V-Charge system only needs a passive bypass valve that can significantly reduce the control strategy calibration efforts and cost.

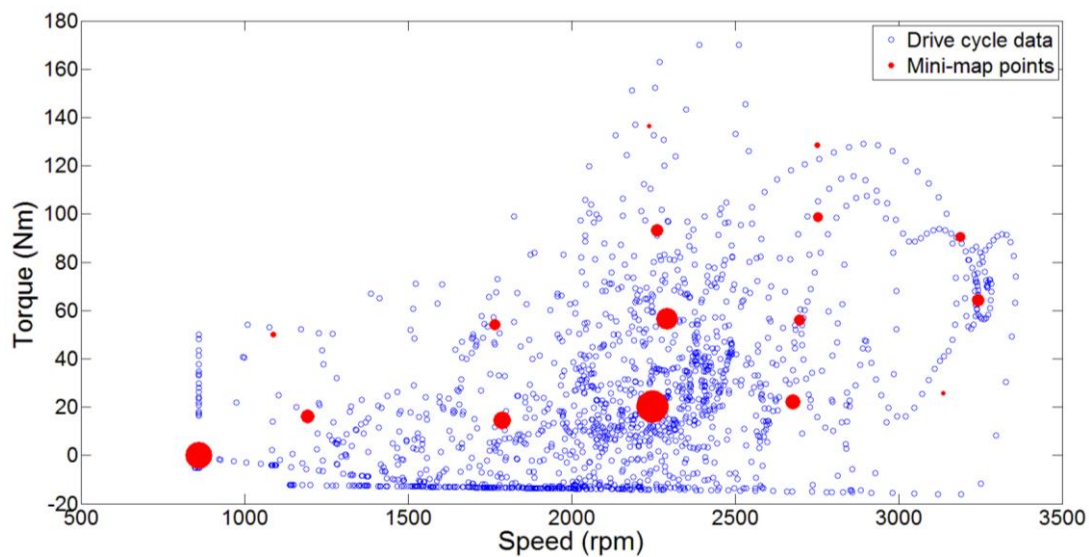


Figure 16. WLTP driving cycle with Minimap

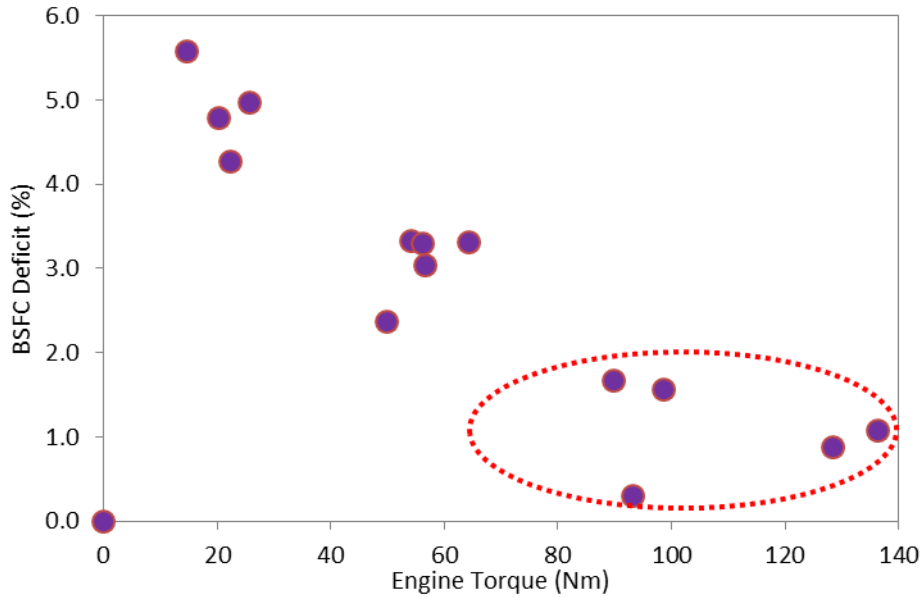


Figure 17. BSFC deficit of V-Charge vs Turbo only for WLTC test points

Transient performance:

Table 1: Control strategy calibration

VC operation \ VC rate	Long	Medium	Short
Fast	1	2	3
Medium	4	5	6
Slow	7	8	9
Turbo-only			

After determining the boost pressure split and populating the control map, some transient tests were conducted, with and without the assistance of the V-Charge system. **Figure 18** to **Figure 20** show the transient torque performance at 1100RPM, 1500RPM and 2000RPM respectively, under four distinctive control calibrations.

There are two parameters that are used to define the V-Charge speed trajectory in a transient: V-Charge speed change rate and V-Charge operation period. It might be noted that unlike the V-Charge speed change rate which could easily be defined as a rate limiter in the control strategy, the V-Charge operation period has to be determined by the transient behaviour of the turbocharger (thus an empirical turbocharger plant has been modelled). By advancing or delaying the turbocharger transient model, different V-Charge operation period could be defined. In order to illustrate how different control calibrations work, test 1, 4, 6 and turbo-only in **Table 1** corresponding to ‘Fast VC with

dip’, ‘Overlength VC operation with overshoot’, ‘Best time to torque’ and ‘Turbo only’ were shown in the following.

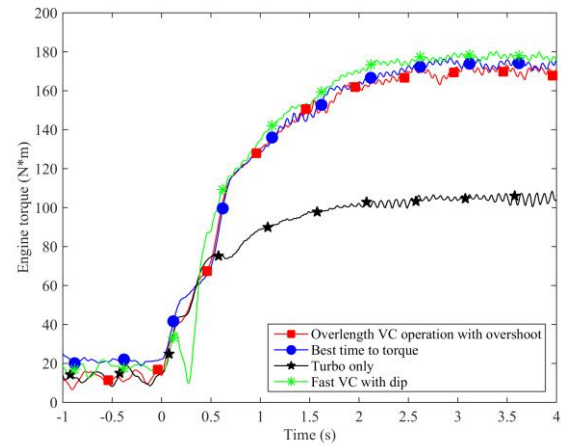


Figure 18. Transient torque performance at 1100RPM

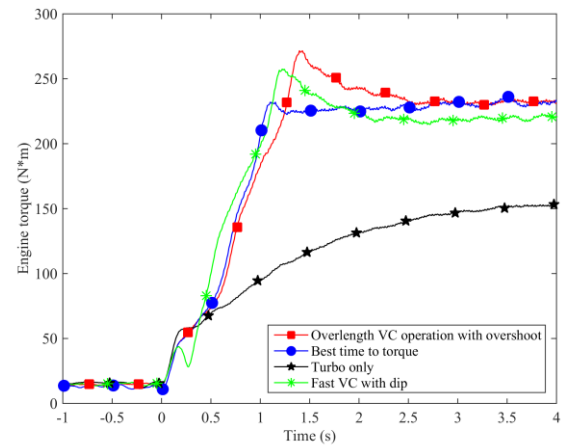


Figure 19. Transient torque performance at 1500RPM

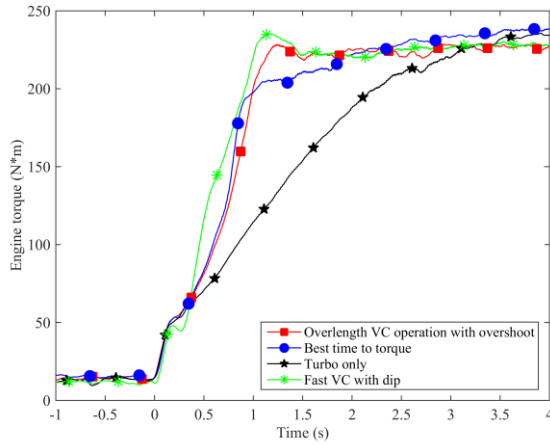


Figure 20. Transient torque performance at 2000RPM

It can be seen that compared to the turbo-only case, the system fitted with the V-Charge system not only has the capability to enhance the final engine torque, but also characteristics significantly improved time-to-torque performance, although different control calibrations will result in different torque trajectory. In order to understand the interactions between the two boosting systems and the engine itself, a more detailed illustrations at a fixed 2000RPM tip-in from 10% pedal position was shown in **Figure 21**.

From **Figure 21 (a) (b)** it can be seen that the V-Charge system, compared to the turbo-only counterpart, can improve the time-to-target performance by approximately 70%, making it behave more like a naturally aspirated engine. Compared to the other calibrations, the case with the fastest V-Charge speed change rate characteristics an engine torque dip during the first phase of the transient which would adversely influence the vehicle's driveability.

If the 'Overlength VC operation with overboost' case and the 'Best time to torque' case have been compared (see from **Figure 21 (a) (d)**), it can be seen that they have broadly the same characteristic when the V-Charge is ramping on but the latter one is ramping off earlier. The result is a faster time to torque behavior and no apparent overshoot. This suggests that there must be a point in the ramp where the V-Charge takes more than it gives back during the end of the transient, due to the fact that mass air flow is increasing as manifold pressure builds so V-Charge uses more power to maintain pressure.

Figure 21 (c) shows the turbocharger compressor behavior during the transient. During the first phase of the tip-in, the turbocharger compressor pressure ratio of the 'Fast VC with dip' case was slightly below 1, due to the fact that the supercharger accelerates much faster than the turbocharger compressor (thus generating higher pressure between the two boosting systems). This 'dip' pressure then helps the acceleration of the turbocharger compressor and makes its pressure ratio higher than the other two V-Charge control calibrations and the turbo only setting. The supercharger's capability to enhance the turbocharger's power was also seen between the 'Best time to torque' case and the 'Overlength VC operation with overshoot' case, where the turbocharger speed and pressure ratio was increased with the prolonged operation of the V-Charge system during the end of tip-in.

The behavior of the V-Charge system can be seen in **Figure 21 (d) (e) (f)**. Compared to **Figure 21 (c)**, the boost split between the supercharger and the turbocharger during a transient can be illustrated: the supercharger pre-boost the engine at the start of a transient and then hands over the boost to the turbocharger while the turbocharger spools up.

It should be noted that the objective of this project is to demonstrate the V-Charge's feasibility as an alternative solution to achieving a highly downsizing concept, thus the tuning of the control strategy which includes the refinement of the feedforward and feedback control is out of scope of this paper. The performance of the V-Charge system could be further improved if a considerable effort of calibration was conducted.

Also it might be worth noting that the optimized V-Charge transient speed trajectory from different engine load should be set differently and the trend is that for a higher starting engine torque, a slower V-Charge speed acceleration rate needs to be used, in order to avoid engine torque dip. This is mainly due to the fact that at low engine torque, after the tip-in, the engine torque that is used to accelerate the supercharger could be offset by the fast operation of the throttle opening, while for a higher engine load, especially above the naturally aspirated region, the torque extracted from

the engine to accelerate the supercharger cannot be offset by the relatively slower operation of the turbocharger and an engine torque dip will be felt if the same supercharger acceleration rate is set.

Figure 22 to Figure 24 show the turbocharger and the V-Charge compressor operating points in the fixed engine speed tip-ins from 10% pedal position for the 'Best time to torque' case. As discussed above, for low engine speeds, the V-Charge system not only accelerates the response behaviour of the turbocharger but also facilitates a higher total boost pressure, extracting higher engine torque than that could be achieved with only the turbocharger. On the contrary, at higher engine speeds, the aim of the V-Charge is mainly to improve the turbocharger response and will return to 'idle' in order to reduce the fuel consumption and avoid breaking the cylinder pressure or intake system limits.

Compared with the fixed-ratio positive displacement solution, due to the capability to be constantly connected with the engine's crankshaft, the V-Charge system does not need a clutch resulting in significantly improved NVH performance during a transient. Also the necessity to disengage the supercharger for the fixed-ratio positive displacement configuration at higher engine speed (due to over-speeding) also affects the driveability consistency. Last but not least, the characteristics of the positive displacement compressor make it mandatory to include a noise attenuation device which is not necessary for the V-Charge system.

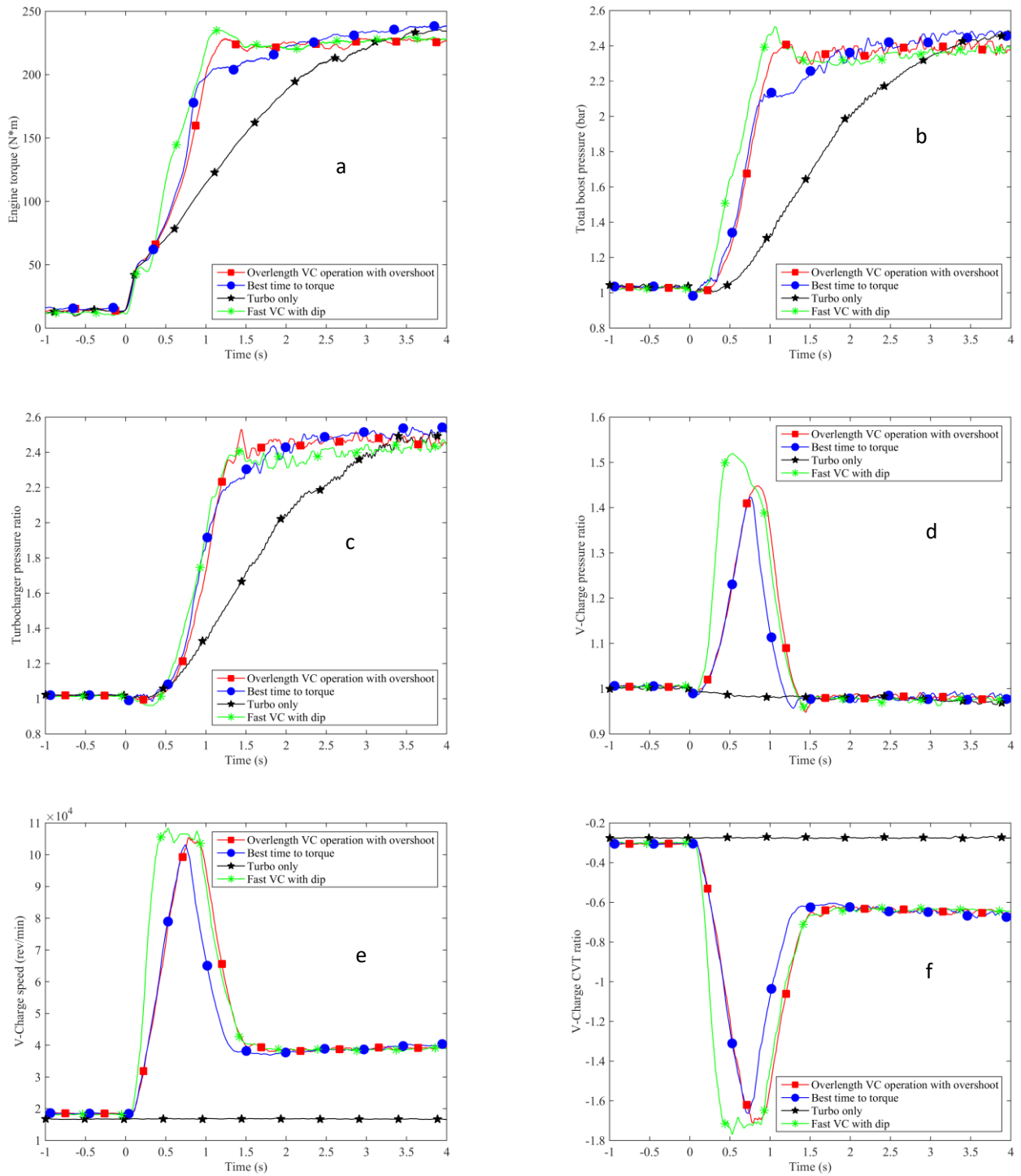
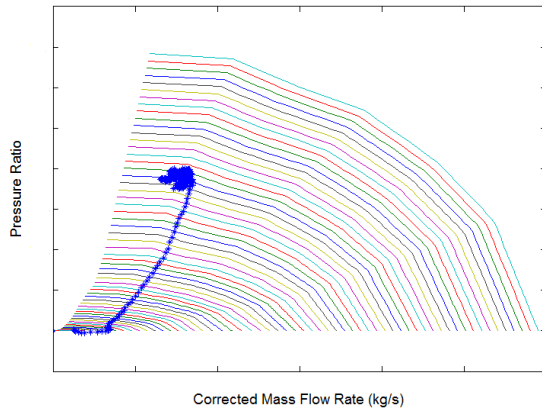
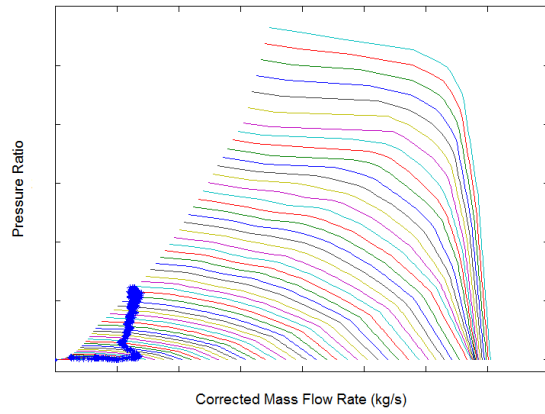


Figure 21. Transient trajectories at a fixed 2000RPM from 10% pedal position

(a) Engine torque; (b) Total boost pressure; (c) Turbocharger pressure ratio; (d) V-Charge pressure ratio; (e) V-Charge speed; (f) V-Charge CVT ratio

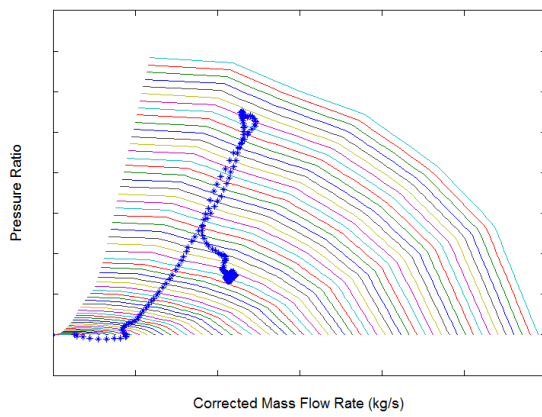


(a)

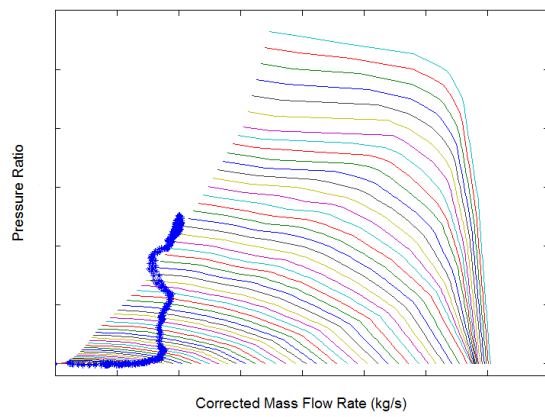


(b)

Figure 22. V-Charge compressor (a) and turbocharger compressor (b) operating points at a fixed 1100RPM from 10% pedal position

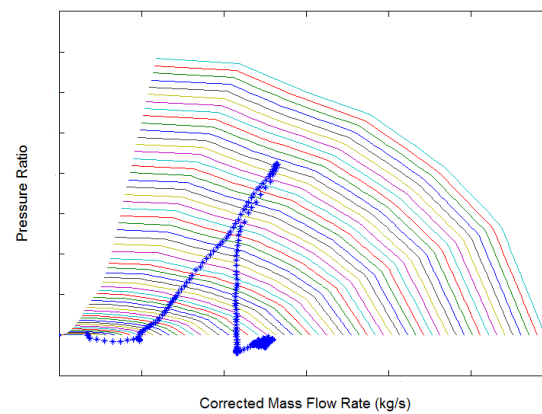


(a)

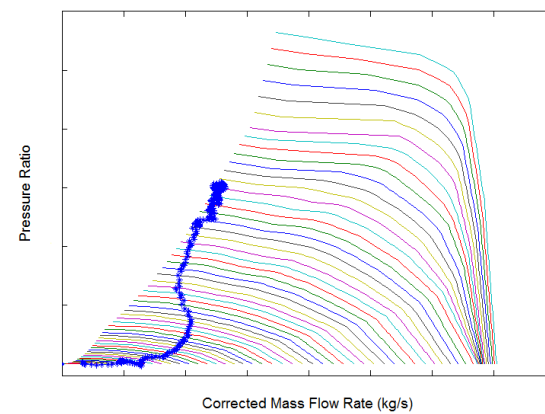


(b)

Figure 23. V-Charge compressor (a) and turbocharger compressor (b) operating points at a fixed 1500RPM from 10% pedal position



(a)



(b)

Figure 24. V-Charge compressor (a) and turbocharger compressor (b) operating points at a fixed 2000RPM from 10% pedal position

8 Discussions

Driving cycle fuel efficiency improvement by further downsizing and down-speeding

From the test data above, it can be seen that both the steady-state full load performance and the transient behaviour for the V-Charge system have been improved significantly compared with the counterparts with only turbocharger. This indicates that the same volume engine with the V-Charge system fitted can drag a larger vehicle, realising downsizing to enhance fuel efficiency. In the meantime, due to the faster transient behaviour, an optimized transmission gear ratio or shifting strategy could be implemented, achieving down-speeding in order to improve fuel efficiency, shifting frequency and driveability [33-35].

The trade-off between part load BSFC and transient performance

For a positive displacement device, a larger pulley ratio will enhance the engine's transient response while compromising some of the partial load BSFC especially at just boosted condition. In addition, a larger pulley ratio could also make the compressor over-speed at lower engine speeds, thus affecting the driveability consistency at high engine speeds.

For a variable drive centrifugal supercharger system, the choice of the drive ratio is also important. A larger drive ratio will, like the positive displacement counterpart above, improve the time-to-torque transient response. However, this will also degrade the part-load fuel efficiency. It might also be noted that if the drive ratio was chosen to be too high, the 'wind milling' might not be working at all.

The potential benefit of Miller cycle for V-Charge system

Miller cycle, usually achieved with an early or late intake valve closing, can achieve a longer expansion stroke than compression stroke, thus improving the engine's thermodynamic efficiency [36-37]. In addition, for a gasoline engine, at part load Miller cycle can

improve the fuel economy due to the reduced pumping losses and for the high load operation, the lowered end-of-compression temperature and pressure for the Miller cycle are able to enhance its anti-knock performance. However, a higher pressure ratio boosting system is required to regain the lost volumetric efficiency and maintain the target performance. The V-Charge system with its superior low-end torque and significantly improved transient behaviour may introduce more flexibility for optimizing the whole engine system with a Miller cycle concept, thus further enhancing its fuel economy.

Further work & Outlook



Figure 25. V-Charge demonstrator vehicle



Figure 26. The V-Charge engine under the bonnet

Following the verification of V-Charge benefits at the engine level, the engine with the original performance target will be fitted into a C-segment vehicle (the same as the baseline engine, show in **Figure 25** and **Figure 26**) and will be re-tuned in order to give a superior driveability. It is expected that the performance of this V-Charge demonstrator is significantly better than the baseline in terms of acceleration time, maximum speed, etc. The down-speeding strategy (including transmission shifting frequency) on the driving cycle

fuel economy is also planned to investigate.

After that the V-Charge system will be upgraded with a higher engine performance (see the stretched target in **Figure 13**) and fitted into a D-segment vehicle. This corresponds to a 33% engine downsizing and a similar procedure as for the C-segment vehicle mentioned above will be conducted to show the performance and fuel economy improvement against the standard vehicle.

Later in the continuation of this project, the V-Charge performance on a diesel engine especially on the aspect of optimizing air fuel ratios will be implemented and the results of that will be published in due course.

9 Conclusions

Mechanically supercharging a passenger car engine is considered to be an alternative or a complementary approach to enable heavy downsizing to be carried out. The V-Charge system, with the capability to enhance low-end torque, improve the transient driveability and reduce the low-load parasitic losses, is deemed to be a potential solution to address the fuel efficiency and driveability issues facing passenger car engines. After investigating, in both simulation and experiment, the V-Charge system on a Ford 1.0L GTDI, the following conclusions are drawn:

1: At steady-state, the V-Charge system is able to significantly enhance an engine's low-end performance without crucially affecting its low load fuel efficiency. Further downsizing or Miller Cycle enabled by the enhanced engine performance could be used to mitigate the fuel penalty that is caused by the parasitic losses.

2: A ready-to-use control strategy at the engine's level has been built and calibrated using the steady-state engine test data. The influence of different control calibrations on the engine performance has been found and will guide the later control calibration tuning at the vehicle's level.

3: In transient, with its wide ratio spread of 10:1 and rapid rate of ratio change, the V-Charge system can

achieve significantly better transient performance in terms of time-to-torque compared with the turbo only configuration, and of NVH behavior in comparison to the fixed-ratio positive displacement counterpart with a clutch. Re-optimizing the transmission gear ratio or the shifting strategy, realizing further down-speeding, could further improve the engine's fuel efficiency in a real driving cycle, while maintaining a good driveability.

4: The V-Charge demonstrator vehicle is expected to either achieve the goal of improving the vehicle's performance or enable an aggressive downsizing to achieve superior fuel economy.

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Abbreviations

BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
CCP	CAD Calibration Protocol
CVT	Continuously Variable Transmission
ECU	Engine Control Unit
FEAD	Front End Accessory Drive
GTDI	Gasoline Turbocharged Direct Injection
IMEP	Indicated Mean Effective Pressure
NVH	Noise Vibration and Harshness
OEM	Original Equipment Manufacturer
PR	Pressure Ratio
TSI	Twin-charged Stratified Injection
VC	V-Charge
WLTP	Worldwide Harmonized Light Vehicle Test

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